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Arthur D. Little, Inc.
CAMBRIDGE 42, MASSACHUSETTS

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Copy No. 9

FINAL REPORT ON

DESIGN AND DEVELOPMENT OF

EXPERIMENTAL MODEL CONDENSING UNIT

To

Contracting Officer
Boston Ordnance District
Contract No. DA-19-020-ORD-84

C-58318

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ARTHUR D. LITTLE, INC.
Cambridge 42, Massachusetts
February 3, 1953

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Arthur D. Little, Inc.

RESEARCH - ENGINEERING - INVESTIGATION

CAMBRIDGE 42, MASSACHUSETTS

February 16, 1953

Boston Ordnance District
Army Base
Boston 10, Massachusetts

Attention: Contracting Officer

Dear Sir: Contract No. DA-19-020-ORD-84

We are submitting herewith nine copies of our Final Report on the above contract for "Design and Development of Experimental Model Condensing Unit."

This is the second contract completed by Arthur D. Little, Inc., on a general project directed toward elimination of fog produced by internal combustion engines in polar regions.

This work under DA-84 comprises a portion, only, of the work originally planned under Phase II of the general project.

As stated in our Final Report, the Contractor recommends that the second portion of Phase II be undertaken. Inasmuch as this entire assignment was initially of an urgent character, it is hoped that immediate consideration be given to our recommendation and steps taken to secure the necessary appropriation.

This recommendation is based on the Contractor's opinion that the assignment under subject contract has been successfully completed and there is every indication that continuing work with full-scale models would be of substantial benefit to the Government.

Respectfully submitted,

Arthur D. Little, Inc.,

1. SUMMARY

DEFINITION OF BROAD PROJECT

The present investigation has been one portion only of a broad scope development project. The purpose of the broad project has been to determine and develop the best means for preventing the formation of fog resulting from operation of internal combustion engines in polar regions.

SCOPE OF PRESENT INVESTIGATION

The scope of the present investigation has been the development and construction of a small model prototype unit based on the findings in an earlier phase of the over-all project. Tests were to be performed with this unit using a cold chamber to simulate polar conditions. Because of the cold chamber limitation, the unit was to be designed to operate with a stationary gasoline engine of about 1 1/2 horsepower.

RESULTS OF PRESENT INVESTIGATION

From fundamental psychrometric data a general theory of operation was developed defining the limiting requirements for any successful system.

This theory was then proven to be correct by construction and test of prototype units meeting these requirements. Tests on the final unit were performed at various cold chamber temperatures from 0°F down to -70°F.

The unit was found to be effective in preventing the formation of fog under all of the conditions tested, down to -70°F, so far as can be determined within the limitations of operation with a cold chamber.

The total volume of surfaces of the final arrangement chosen was .087 cu. ft.; weight 3.6 lbs.; blower power .00265 theoretical air horsepower.

Transposed to a 100 horsepower unit, the indicated size of the parallel exchanger surface sections would be approximately 10" x 12" x 48", which would replace the present muffler. Blower power would be about 0.5 horsepower. This size could be reduced materially at the expense of increased air horsepower.

CONCLUSIONS AND RECOMMENDATIONS

It is concluded that heat exchangers can be constructed to eliminate effectively the formation of fog resulting from operation of internal combustion engines in polar regions, under conditions where no supersaturation of the ambient air is present.

It is indicated that these units will not be of as small bulk as originally expected, but that they still offer the best practical method for fog elimination.

It is concluded that the precise arrangement and final size of a unit for any given application cannot be predicted by application of a general equation, since they will depend on such factors as the geometry of the space available, the relative importance of exchanger weight and bulk versus power consumption in the particular instance, and other factors of like nature.

It is recommended that the contracting office authorize the next step in the over-all program (Step Two of Phase II); namely, to develop and construct a full-scale prototype model. This full scale model would be constructed to suit the requirements of some particular vehicle or unit designated by the Boston Ordnance District, preferably of about 100 horsepower.

SUMMARY OF WORK DONE

Efforts were made to purchase suitable condensing surfaces from regular manufacturers of such equipment, these surfaces to be approximately of the form determined during Phase I.

A number of possible arrangements to obtain the necessary condensing conditions were investigated and several preliminary models made and tested. The factors of practical suitability for the service intended and for quantity production were considered. Finally, the most reasonable design was chosen and a model constructed. Since the model was for only a 2 horsepower engine, one of the chief considerations was to choose the dimensions of the model so that test results from it could be conveniently transposed to apply to a considerably larger unit.

2. BACKGROUND AND DETAILED DESCRIPTION OF PROJECT

REQUIREMENTS

The purpose of the over-all project has been to determine and develop the best means for preventing the formation of fog resulting from operation of internal combustion engines in polar regions.

At the outset, it was established that the means developed should possess the following general Military Characteristics:

1. Simplest possible construction adaptable to mass production.

2. Minimum practicable weight and bulk.

3. Minimum practicable maintenance requirements.

4. Capable of withstanding the shock and strains attendant the operation of the vehicle over all types of arctic terrain.

5. Capable of installation without special tools and in minimum practicable time, by third echelon maintenance personnel.

6. Capable of being stored, transported and issued as a component of the arctic kit (for winterization of vehicles for temperatures from -40° to -65°F).

In addition to the above characteristics, the device should not appreciably affect the performance of the engine to which it is attached, or require an excessive amount of power for its operation.

SUBDIVISIONS OF PROJECT

It was originally expected that this project would be conducted in four phases, as follows:

Phase I - Study of conditions under which automotive engines produce ice fog; methods of preventing or minimizing such ice fog; preparation of military characteristics for an anti-ice-fog device for automotive engines.

Phase II - Design and construction of prototype devices.

Phase III - Engineering tests at Aberdeen Proving Ground or other suitable facility.

Phase IV - Service tests by Army Field Forces and other agencies concerned.

SUMMARY OF PHASE I

The work by Arthur D. Little, Inc., on this phase was done under Contract No. W19-020-ORD-6522. A final report was submitted December 15, 1949, entitled, "A Survey of Exhaust Gas Water Vapor Eliminator Equipment." This was a theoretical study and review of data in order to recommend means for eliminating the fog formations which accompany operation of internal combustion engines in polar regions.

A survey was made of all of the possible means for accomplishing this result, and it was concluded that heat exchangers offered the best practical solution to the problem. Two types of heat exchangers were examined in detail; a perforated plate type, and a plate-louvered fin type, and it was concluded that the latter would come nearest to meeting the requirements. A particular pattern of this type was chosen for specific calculations. This pattern was one made by Harrison Radiator Division of General Motors Corporation and described in Research Memorandum No. 2-46 Navshps (250-338-3) dated 1 July 1946 entitled "Gas Turbine Plant Regenerator Surfaces--Basic Heat Transfer and Flow Friction Data." Numerous types of surfaces are described in this report; the surface designated as Type "C" was the one chosen in this case.

For the calculations, a 100 horsepower engine was chosen as an example, and the use of brass and aluminum as the heat exchanger metals was assumed; brass for the wet end and aluminum for the dry end. A series of curves was developed to show the relationships of plate weight, exhaust gas pressure drop, and cooling air blower horsepower for various assumed mass velocities and ratios of cooling air flow to exhaust gas flow. These curves were based on the assumed requirement of cooling the exhaust gases from 1000°F to 40°F, with cooling air at -40°F. It was assumed, subject to later experimental proof, that if the exhaust gases were cooled to 40°F sufficient condensation would have taken place so that the remaining moisture vapor would not create a visible fog.

Finally, it was recommended that the contracting office initiate a program for the design, construction and testing of a model, and finally prototype engine fog elimination units employing plate-louvered fin crossflow type heat exchangers.

SCOPE OF PHASE II

Design and construction of Prototype Devices -- The proposal by Arthur D. Little, Inc., to undertake the work of Phase II was subdivided into two steps.

It was proposed that Step One would provide for the development and construction of a small model. The results of this work could then be fully evaluated before proceeding to Step Two. This second step would comprise the development and construction of a full scale prototype unit suited to a specific military vehicle or piece of equipment.

A contract for Step One of this Phase II was received by Arthur D. Little, Inc., on April 3, 1951, which is the present contract (DA-19-020-ORD-84) entitled "Design and Development of Experimental Model Condensing Unit."

3. PRELIMINARY WORK DONE UNDER PHASE II

PROCUREMENT OF EXCHANGER SURFACES

As a first step, procurement of a suitable test model heat exchanger of the plate-louvered fin type was undertaken.

In presenting specifications to the manufacturers the requirements for a 100 horsepower engine were given, with requests for their estimate on the most suitable exchanger design. The ultimate requirement of suitability for mass production was particularly stressed.

Six manufacturers in the field of plate-type exchangers were contacted as follows:

The Harrison Radiator Division
General Motors Corporation
Lockport, New York

The Trane Company
LaCrosse
Wisconsin

The Elliott Company
Jeannette
Pennsylvania

The AiResearch Manufacturing Company
Los Angeles
California

The Air Preheater Corporation
Wellsville
New York

Clifford Manufacturing Company
Grove Street
Waltham, Massachusetts

Only one of these, Harrison Radiator Division of General Motors Corporation, was able to produce the exact pattern recommended as best in the Phase I report. Three other companies, Trane, Air Preheater, and Clifford, suggested designs of their own which in their opinion would accomplish the same purpose. The other two were

unable to offer exchangers suitable for the purpose. The Harrison Radiator Division accepted an order placed June 14, 1951 with an agreed delivery date of "approximately sixty days." The Trane Company, one of the other companies suggesting a design of their own, also made a firm quotation which was accepted and an order placed June 25, 1951. The other two companies that had suggested designs withdrew their offers.

TEST EQUIPMENT

While waiting for arrival of exchangers, the necessary test equipment was assembled and set up.

A diagram of the installation as originally planned is shown in Figure 1.

A list of the equipment is given in TABLE I.

Using a cold chamber that was available at Massachusetts Institute of Technology, the equipment was set up substantially as outlined in Figure 1, except the twelve-point potentiometer was not available in time so portable instruments were employed. It was found impractical to put the exchanger inside the cold chamber so it was mounted outside and connected with insulated ducts. In addition to the small flowmeter for measuring fuel input, the gasoline tank was mounted on scales for periodic weighing. A photograph of the actual installation is shown in Figure 2.

PRELIMINARY TESTS ON SIMPLE CROSSFLOW UNIT (TRANE UNIT)

It was soon found that merely cooling the exhaust gases to 40°F did not lower the moisture content sufficiently to prevent fog formation when injected into the cold chamber. Moreover, in order to obtain the 40° exit temperature, such a low metal temperature was necessary with the crossflow arrangement that the exhaust passages soon froze up.

It was also found that the limited size and capacity of the cold chamber at Massachusetts Institute of Technology was insufficient for reliable tests of fog formation. The cooling air temperature could not be maintained at the desired level long enough to stabilize operating conditions.

After completion of these tests, the Harrison exchanger, the one actually recommended in the Phase I report, had still not been received. It was found that the parts for it had all been fabricated, but not yet assembled. In view of the change in requirements which would necessitate rearrangement of surfaces to provide for a parallel flow section, Harrison was requested to ship the unassembled parts rather than risk another long delay in trying to obtain delivery of complete units. These parts were received late in July, 1952, and were used to fabricate the final prototype units.

INVESTIGATION OF OTHER COOLING ARRANGEMENTS

Since tests with the Trane exchanger had apparently indicated that cooling alone was insufficient, calculations and experimental work were undertaken to determine the best method of meeting the conditions theoretically required.

One method investigated was to employ a two pass counter-current exchanger of considerable length which would cool the exhaust gases in one pass by rewarming them in the other, discharging them at as high a temperature as possible. This section of the condenser would require no cooling air at all, thereby reducing the blower power requirement. The only cooling air necessary would be for a small section, required for removing the heat of condensation of the water vapor. With an assumed exhaust temperature of 1000°F and ambient temperature -40°F as before, calculations indicated that the theoretical exhaust exit temperature would be 690°F. Attempts were made to check this result experimentally, which was difficult to do because of the necessarily finite size of the cold chamber. By the time conditions were stabilized at the desired values some fog would already be created in the chamber so it was impossible to tell for sure that no additional fog was being created. However, the evidence seemed to indicate that the calculations were correct.

Since one of the most desirable features of any system would be a low power requirement for the cooling air blower, the possibilities of this type of system were investigated further.

A considerable portion of the counterflow section of the exchanger surface for this system would have to operate at fairly high temperatures; therefore it would probably be necessary to use ferrous materials. Since the heat transfer coefficient of ferrous materials would be comparatively low, secondary surfaces would not be very efficient, so they were eliminated.

SUITABILITY FOR MASS PRODUCTION

Several designs were developed for formation of the plate surfaces which would be suited to low cost mass production and assembly operations. Furnace brazing methods were investigated, and also a method which would require no brazing or welding at all. For this latter, a pattern for cutting and bending the individual plates before assembly was developed which automatically provided for the necessary headering. After being bent to the desired shape, the stack of plates could be assembled into a complete exchanger, including headers, merely by clamping them together.

In order to maintain as light and compact a design as possible the cooler portions of the counterflow section should be aluminum. In this portion secondary surfaces therefore could be used to advantage. However, for the test model this additional complication was avoided, the whole counterflow section being made in one style of construction suitable for the higher temperatures.

As a result of these tests, however, even after allowing for improved efficiencies with a more elaborate surface arrangement, it was apparent that for most applications the larger size and greater weight of such a system as this would be two disadvantages, more than offsetting the advantage of minimum power for the air blower.

4. DESIGN, CONSTRUCTION, AND TEST OF FINAL UNIT

GENERAL THEORY OF OPERATION

The basic principle underlying the operation of any successful unit can be illustrated by use of the conventional psychrometric chart for moist air. On this chart, the abscissae represent the dry bulb temperatures and ordinates represent the percentage moisture content. A simplified version of such a chart is shown in Figure 3. The state of any moist air (emerging exhaust) mixture is represented by point X on the chart. The state of any ambient condition may be represented by point A. For this illustration, point A is chosen <u>on</u> the saturation curve, corresponding to 100% relative humidity.

If a line is now drawn from point A to point X, this line will be the locus of the state points of any mixtures resulting from mixing any ratio of the original moist air (emerging exhaust) with the ambient air. This statement assumes that the temperature resulting from the mixing is a weighted arithmetical means of the two original temperatures. It is therefore only approximately true since the heat content of the water vapor alters the final temperatures slightly. However, within the range of temperatures and moisture ratios under consideration, the error resulting from this assumption is negligible. There is also an error involved in applying the psychrometric chart for moist air to the present problem of exhaust gas mixtures, but this also is small.

An illustration of the above statements, taking specific values, is as follows:

State of mixture at pt. X: 3 gr/# @ 50°F
State of ambient air (pt. A): .55 gr/# @ -40°F
Assume that 1 part mixture at pt. X is mixed with 5 parts ambient air (pt. A)

$$
\begin{array}{rl}
-40 \times 5 = & -200 \\
50 \times 1 = & \underline{50} \\
5 + 1 = 6 & 6 \overline{)-150}
\end{array}
$$

Final Mixture Temperature = -25°F

$$
\begin{array}{rl}
.55 \times 5 = & 2.75 \\
3 \times 1 = & \underline{3} \\
 & 6/5.75
\end{array}
$$

Final Moisture Ratio = 0.96 grain/#

From further observation of the chart it will be seen that if a line AB is drawn on the chart through point A, tangent to the saturation curve at this point, this line will define the maximum limit of moisture content for any emerging exhaust mixture leaving the condenser. If its state point falls in the shaded area below line AB, any ratio of mixing with ambient air will not produce fog. If its state point falls above line AB, as at point Y, then the mixtures with ambient air whose state points lie between A and Z will be supersaturated, and fog will be formed.

LIMITATIONS

In the above analysis it is assumed that no conditions of supersaturation exist in the ambient air. It is probable that actual field conditions at times would be such that actual performance would not be strictly according to this principle. However, it is believed that this would be infrequent. There was no indication of effects of this kind during any of the cold chamber tests; these tests in all cases indicated behavior closely in accordance with the predictions of the theory.

DESCRIPTION OF FINAL UNIT

Based on the above theory, a compromise system of cooling and mixing was developed which makes as much use as possible of the large temperature difference available yet eliminates the danger of freezing. To accomplish this, the cooling load was divided between two exchangers as before, but in this case cooling air was used in both of them. A temperature diagram of this arrangement is shown in Figure 4.

The first exchanger was a crossflow type designed to drop the exhaust temperature rapidly to about 300°, the lowest temperature that it was considered safe to attempt in a crossflow arrangement without danger of freezing in the coldest corner.

The balance of the cooling operation was carried out in a parallel flow concurrent type of exchanger, with the cooling air therefore warming up as the exhaust stream was cooled, so that the lowest metal temperature in the exchanger remained above freezing. This lowest metal temperature occurred at the exit, where the exhaust stream had been cooled to slightly above freezing and the cooling air warmed to only slightly below.

The condensing load is carried almost entirely by the second exchanger. The leaving exhaust stream carried with it all of the condensed water, mostly in the form of fairly large entrained droplets. These were removed in a cyclone type separator, the exit gas stream thereafter mixing with the two cooling air streams from the two exchangers before being discharged. This mixing was accomplished by enclosing both exchanger sections and the water separator and trap in a housing with mixing baffles at the outlet. Such a housing further would be necessary in actual service to prevent the exchangers from freezing up when first starting. An exhaust bypass relief valve, installed in the transition duct from the crossflow section to the parallel flow section, was set to open and bypass the exhaust stream into the housing if any abnormal back pressure should develop in the parallel flow section.

After mixing was accomplished, the heat and moisture content of the mixture was then such that the state point was below the limiting tangent line on the chart as described above, and no matter what the subsequent mixing ratio with ambient air, the saturation point was never reached. Fog, therefore, was not produced.

DESIGN OF FINAL UNIT

Design calculations for the final test model are given in Appendix B, pages 1 to 11. The pertinent data are summarized in Figure 5.

DIMENSIONS OF FINAL UNIT

The dimensions of the unit as actually constructed were varied slightly from those of the design, in order to use available heat exchanger surfaces. The completed unit is shown in perspective in Figure 6. Figure 7 is a photograph of the unit while undergoing test.

TEST OPERATION

A general view of the test is shown in Figure 8. The layout is shown diagrammatically in Figure 9, and a list of the equipment used is given in TABLE II. The arrangement was similar to the earlier tests except a separate source of cold air for the cooling blowers was provided. This change was made in order to prevent disturbance of conditions within the cold chamber. In the earlier tests, conditions could not be maintained constant, because air from the cold chamber was used in the exchanger, which soon affected the temperature and humidity conditions within the chamber. With a separate source of cold air for the condenser, this difficulty was overcome.

Two cooling air blowers were used, one on each section of the unit. The blower on the crossflow section was a propeller type with 1/150 horsepower motor. This fan moved about 13 c.f.m. of air at .008" H_2O static pressure. In actual service it is expected that the cooling needs of this section could be met by convection or ram effect.

The cooling requirement of the parallel flow section was met by a small squirrel cage type blower operated by a 6^V DC motor, the speed of which could be varied to maintain the desired 40°F discharge temperature of the exhaust gas stream emerging from the water separator. In a full scale model, this control would be automatic, and would be the only control necessary.

Temperature measurements were made at various points throughout the unit as shown in Figure 10. A thermocouple connected to a twelve-point switch and Brown Instrument Company potentiometer was used at the positions circled, with Weston Instrument Company dial thermometers at all of the other positions.

TEST RESULTS

The results of two of the most significant test runs are shown on the data sheets in the appendix. Run No. 22 was made with cold chamber temperature at -46°F. Run No. 24 was made at -70°F. Although a portion of the exhaust stream was discharging into the chamber steadily for one hour ten minutes and fifty minutes respectively, no fog formation within the chamber occurred during either of these runs, as long as the condensing temperature was maintained near 40°F. At the end of Run No. 22, this temperature (the discharge temperature of the exhaust from the water separator) was permitted to rise gradually above the 40°F level. The chamber immediately began to fog up as soon as this temperature reached 47°F. In summary, 1) the condenser was functioning satisfactorily in suppressing fog so long as the design condition of 40°F exhaust discharge temperature was maintained, and 2) operating conditions had to be controlled accurately; a rather small percentage increase in condensing temperature could not be tolerated.

CONCLUSIONS

Because the unit functioned closely in accordance with predictions of the general theory of operation, it is concluded that this theory can be reliably applied to the construction of a unit for any required application and set of conditions.

The precise arrangement would vary depending upon the specific requirements in each case. For instance, in some cases size may be more critical than power requirement, or vice versa. The final unit constructed was just one possible arrangement which was found to operate successfully. Many other possible arrangements could be made. Besides practical considerations, the only basic requirements are:

1) to cool the exhaust to approximately 40°F without permitting metal temperature within the condenser to fall below freezing temperatures

2) to separate the condensed water from the leaving exhaust stream

3) to provide sufficient heat by mixing the leaving exhaust stream with all (or a sufficient portion) of the cooling air warmed in the cooling process so that the state of the resulting mixture is below a certain limiting range which is determined directly by the ambient conditions

A quick method of determining this range is to make use of a psychrometric chart, erecting a tangent to the saturation curve at its intersection with the ambient temperature. Provided the state of a mixture falls below this tangent, fog will not be produced.

Under usual conditions of operation for all internal combustion engines, sufficient heat is provided by the exhaust stream to meet this condition.

APPENDIX A

SECURITY INFORMATION
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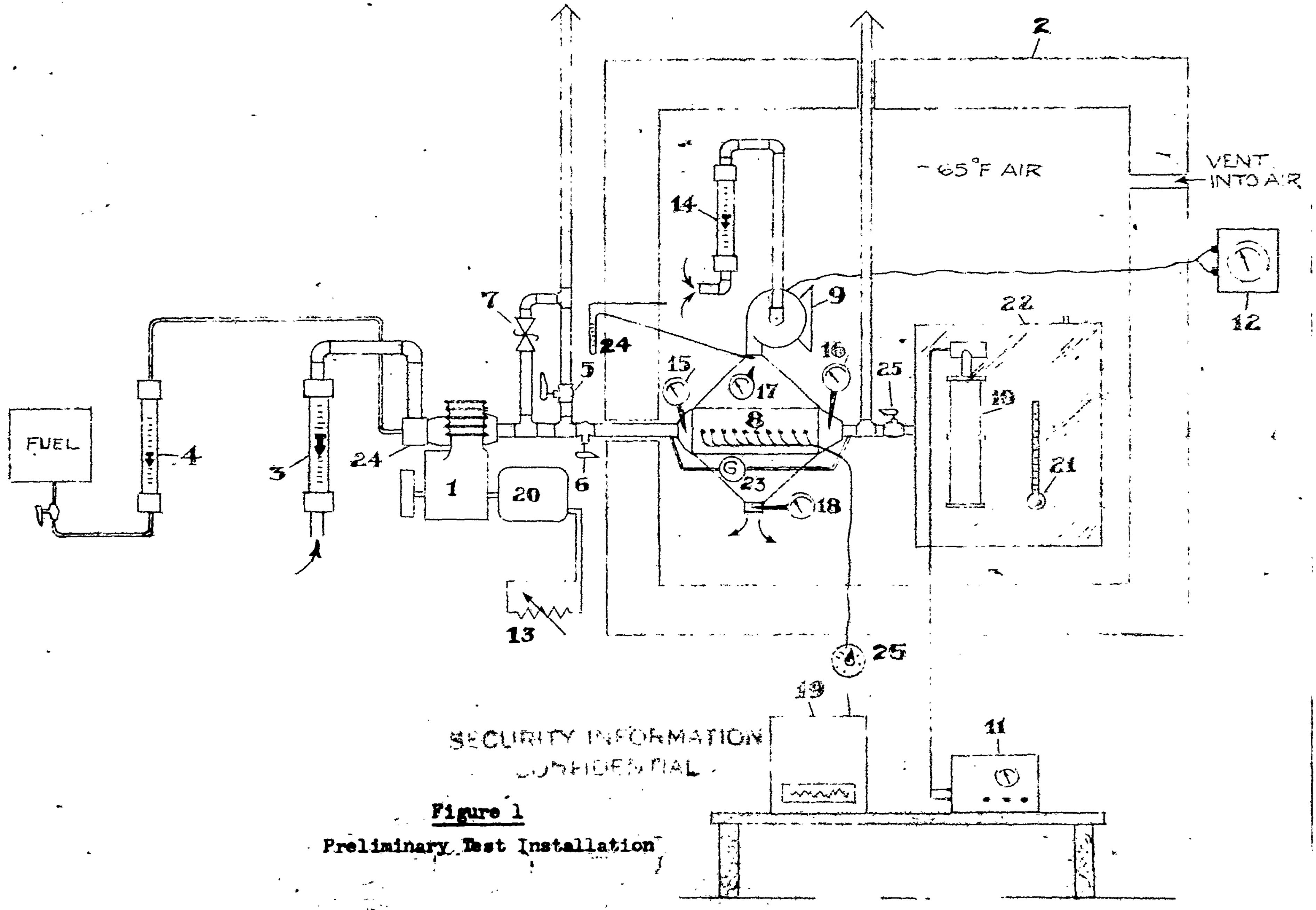

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Figure 1

Preliminary Test Installation

Figure 2

Installation - Preliminary Tests

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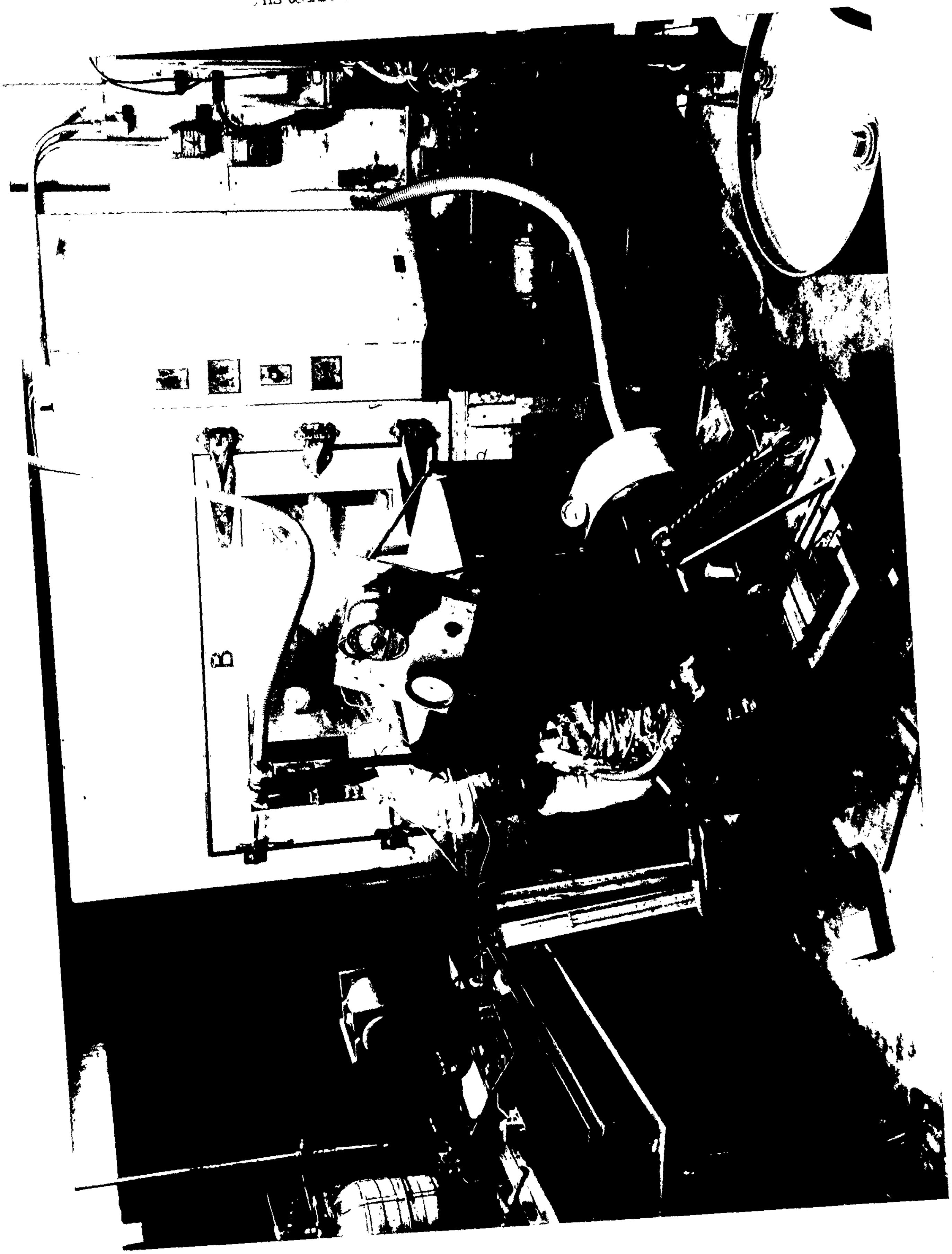


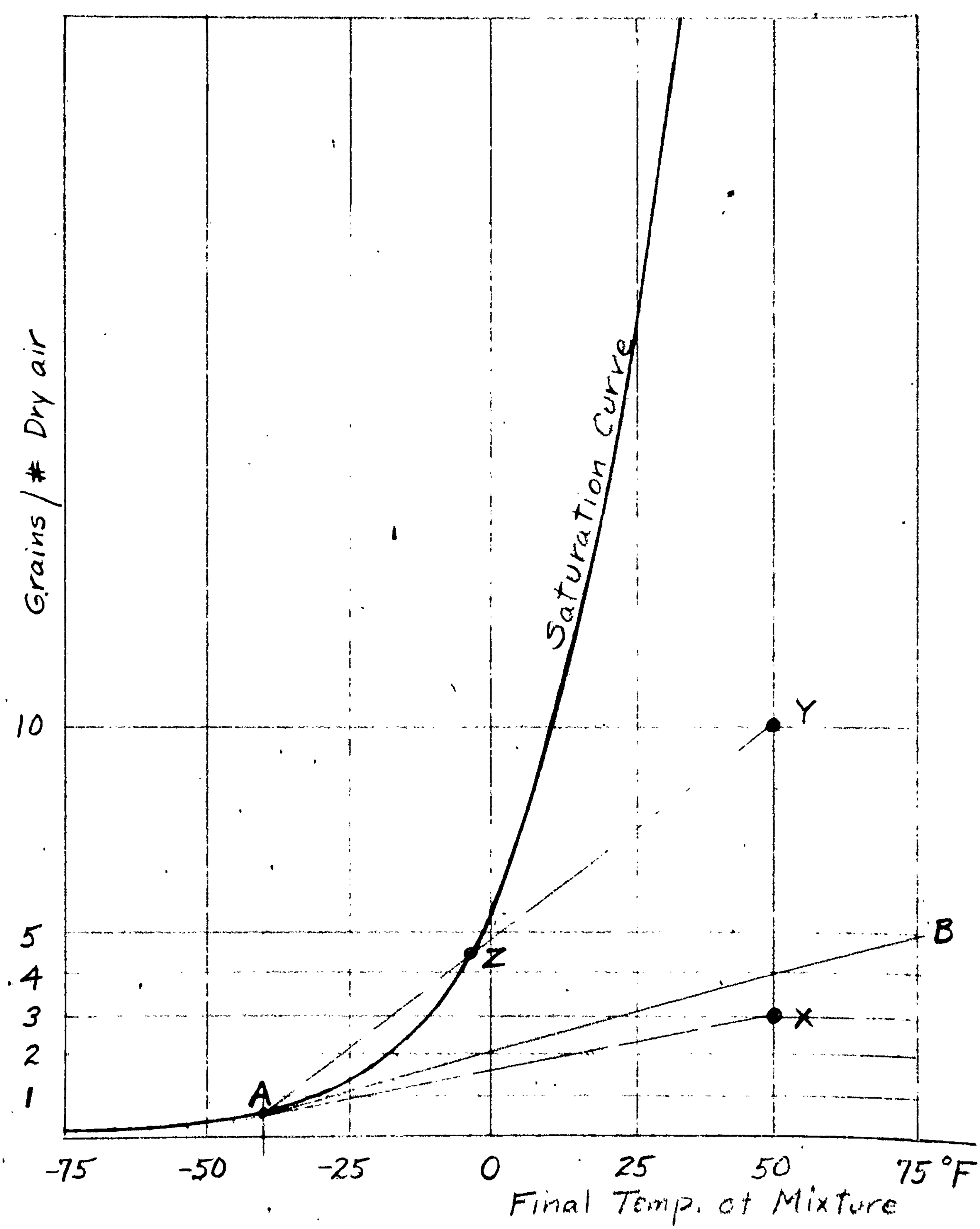
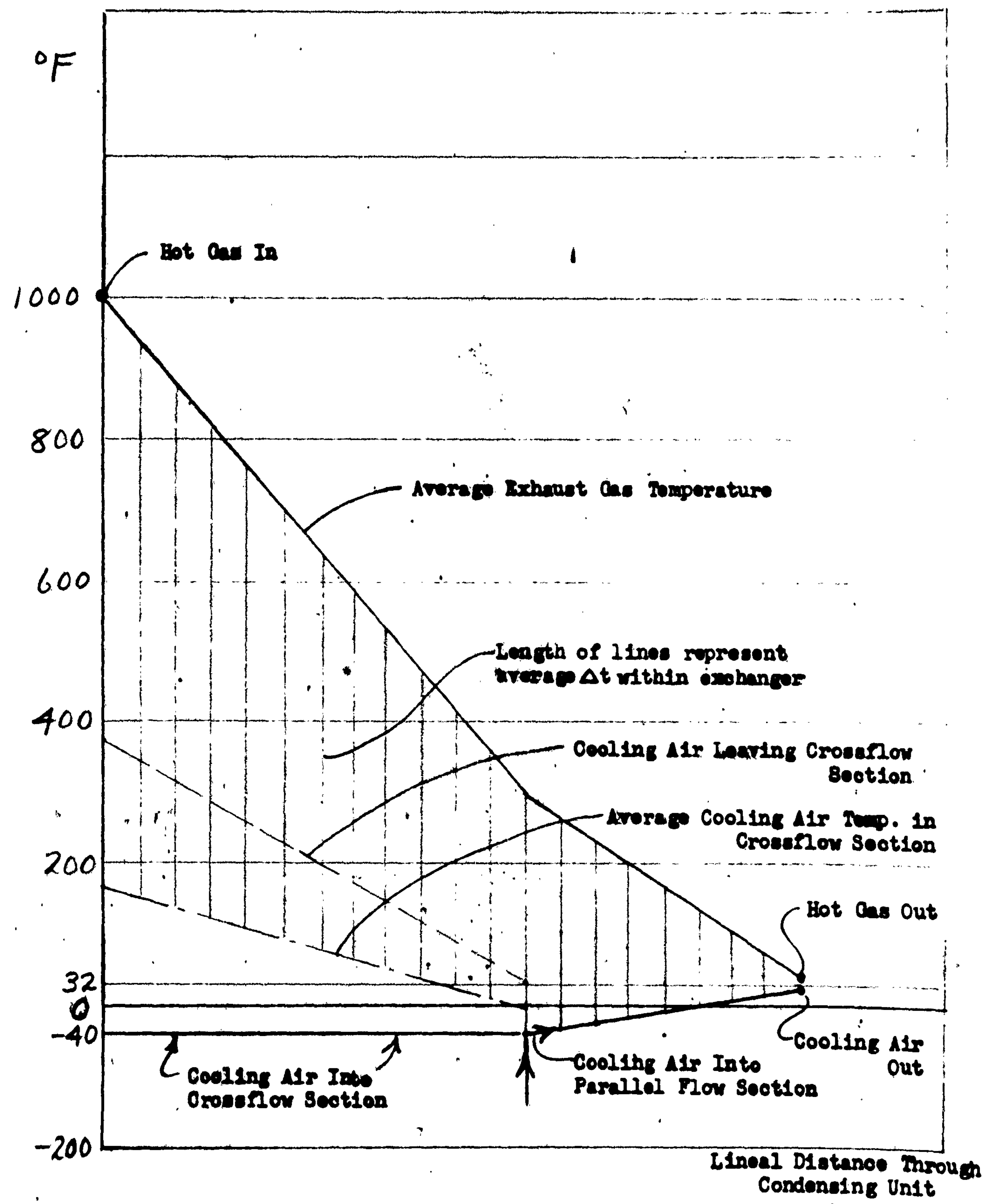


Figure 3

Psychrometric Chart SECURITY INFORMATION



Temperature Differentials - Final Unit
Figure 4

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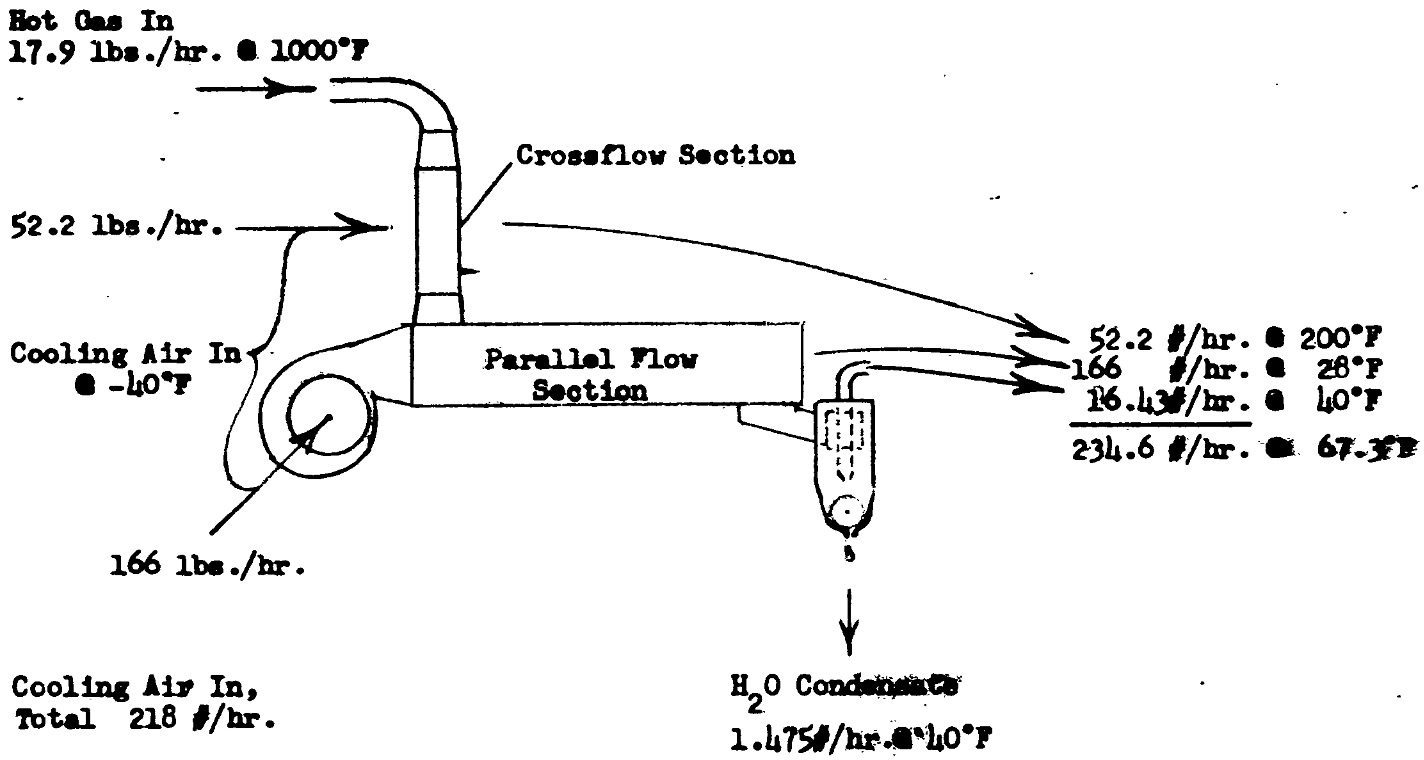

Dimensions of Unit:

	Crossflow Section	Parallel Flow Section
Width of Core	5-1/2"	3"
Height of Core	4"	3-1/4"
Thickness of Length	1-1/2"	12"
Cooling Air Face	5-1/2 x 4"	3 x 3-1/4"
Exhaust Gas Face	5-1/2 x 1-1/2"	3 x 3"
Assembly	Silver Soldered by furnace brazing	Solder Dipped

Core Pattern (both sections)	
Plate Face Separation	0.25"
Fin Spacing	11.1/in.
Plate Metal Thickness	0.012"
Fin Metal Thickness	0.006"
Louver Spacing	1/4"
Plate Material	Inconel
Fin Material	Inconel

Figure 5

Specifications and Design Data - Final Unit

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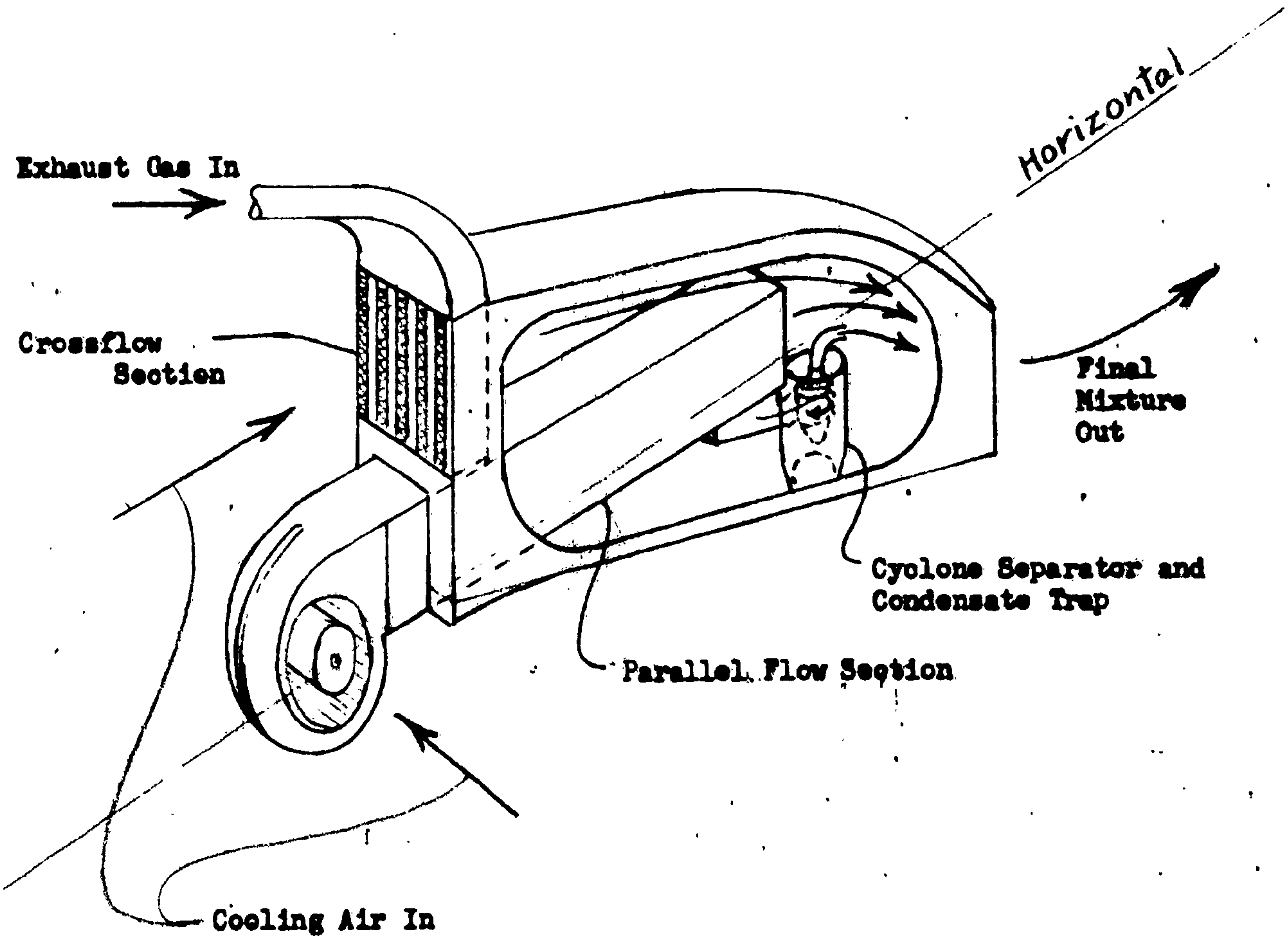


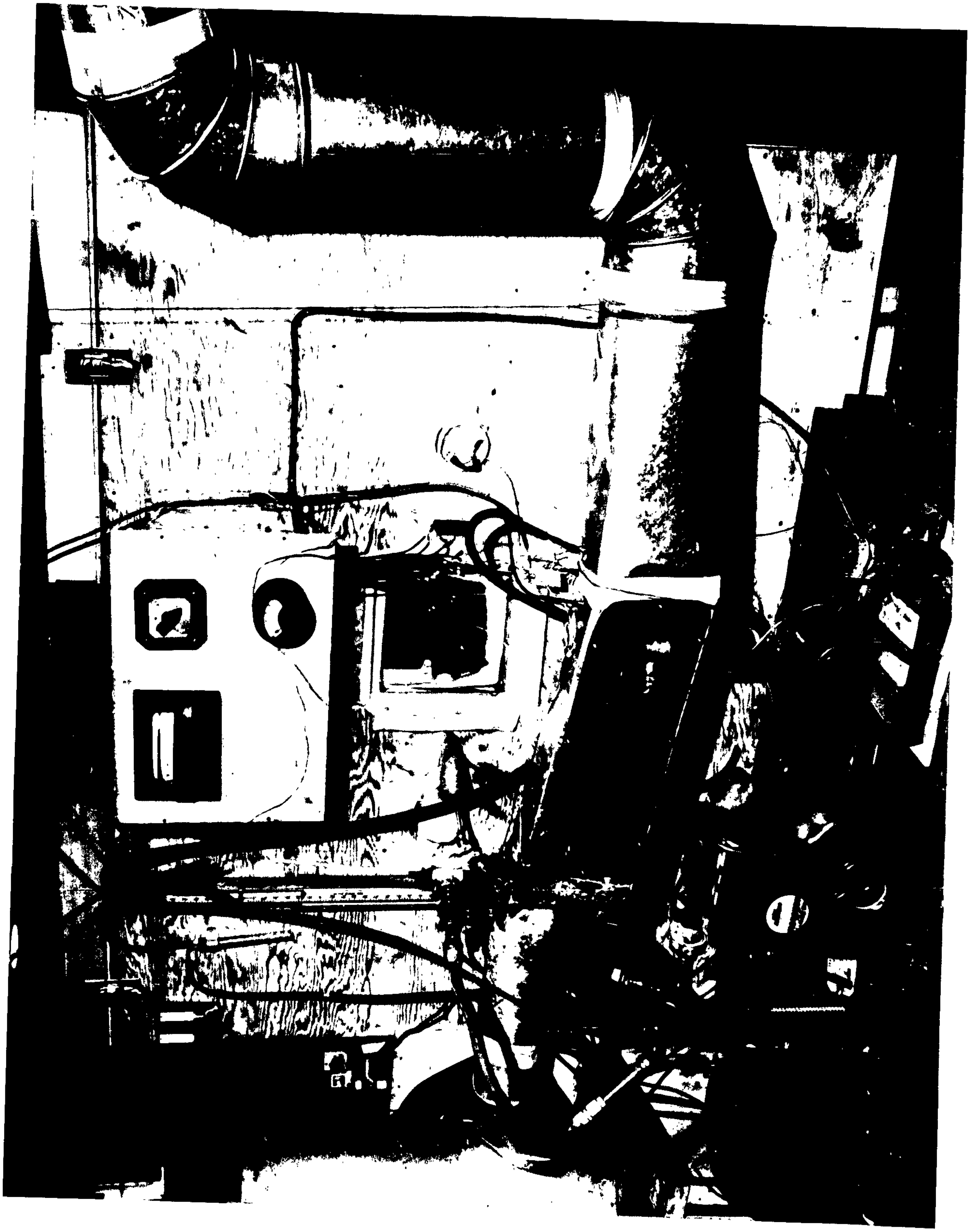
Figure 6

Perspective View - Final Unit

Figure 7
Final Unit Undergoing Test

Figure 8

Installation - Final Tests



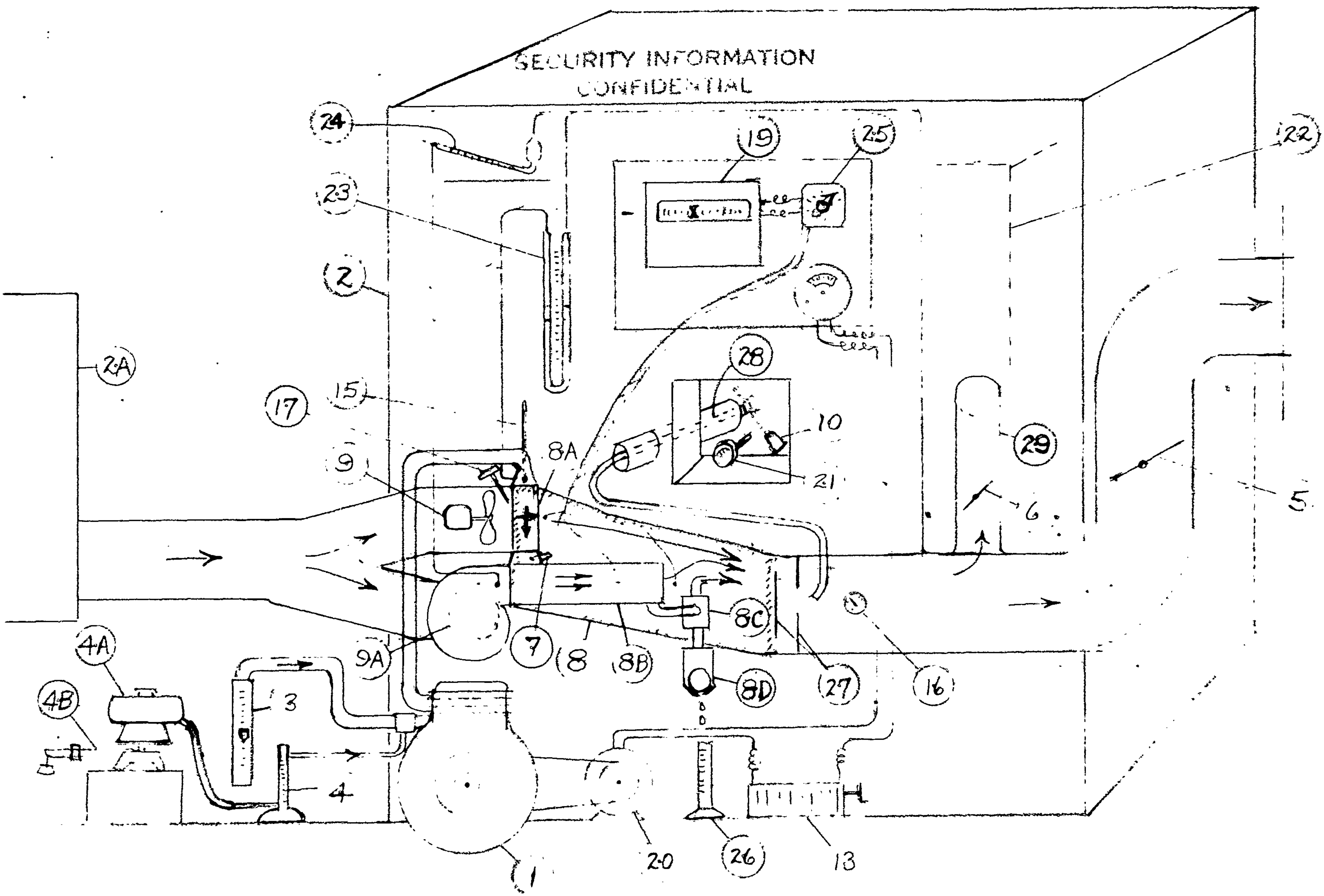


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Figure 9

Diagrammatic Layout of Test Installation - Final Tests

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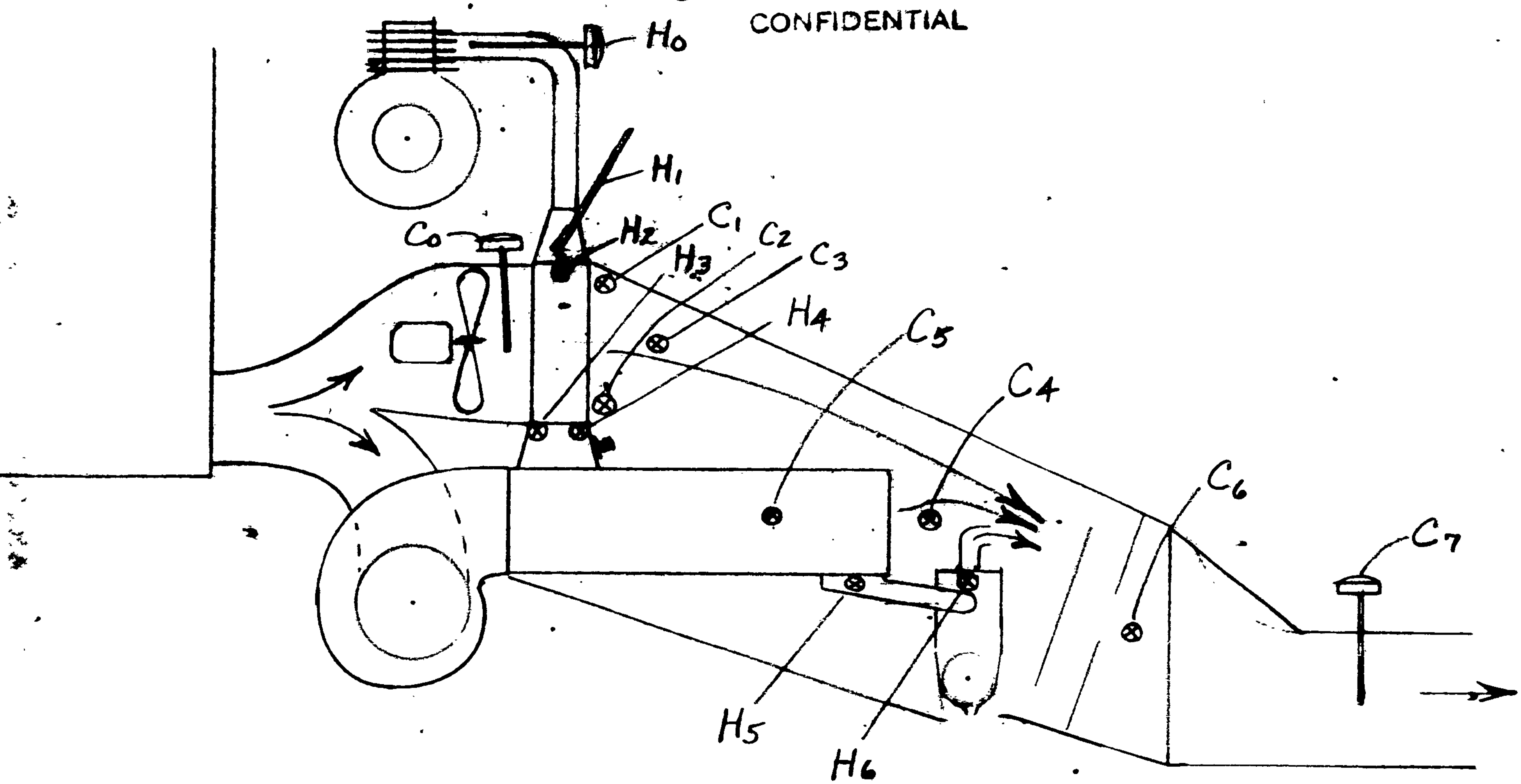

Figure 10

Locations of Temperature Measuring Points - Final Unit

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TABLE I

Test Equipment (Preliminary Tests)

No.	Item	Source	Description
1	Engine	Wisconsin Motor Corp.	Model ABN .85 HP @ 1400 RPM 2.10 HP @ 1800 RPM
2	Cold Chamber	M. I. T.	Refrigerant cooled. Variable to 65°F. 70 cu. ft.
3	Flowmeter for Intake Air	Fischer & Porter	Flowrator Size 5 Fig. #735-BH 10" scale
4	Flowmeter for Fuel	Fischer & Porter	Flowrator Size 07 Fig. #130-BH 150 mm scale
5	Blast gate to vent	Rockwell Mfg. Co.	BT-401
6	Blast gate to test	Rockwell Mfg. Co.	BT-401
7	Exhaust Pressure Safety Valve	J. A. McDonald	3/4" side outlet set 5 psi
8	Exchanger undergoing test	Trane Company	
9	Air blower	Trane Company	8" aluminum squirrel cage type
10	Light Scattering Chamber	---	
11	Penetration Meter	---	
12	Blower Power Meter	---	
13	Variable Resistance	---	Carbon Pile

TABLE I (continued)

14	Flowmeter for Cooling Air	Fischer & Porter	Size 8, Fig. #735-BH
15	Thermometer - Hot gas in	Weston Electr. Inst. Co.	200° to 1000°F
16	" - Exhaust Gas Out	" " " "	-100° to +100°F
17	" - Cooling Air In	" " " "	-100° to +100°F
18	" - Cooling Air Out	" " " "	-100° to +100°F
19	Potentiometer	Brown Instr. Div. of Minneapolis Honeywell Regulator Co.	-50° to =1000°F range
20	Generator	A & J Auto Ignition Co.	6^V DC 40 amp.
21	Thermometer - Inner Fog Chamber	Macalaster Bicknell	-100° to +150°F
22	Inner Fog Chamber	---	12 cu. ft. Glass Enclosed
23	Differential Pressure Gauge Hot Side	U. S. Gauge Company	0 to 10" hg
24	Pressure gauge - Cold Side	Meriam Inst.	" H_20 manometer
25	Selector Switch	Brown Instr. Div. of Minneapolis Honeywell Regulator Co.	12-pt.

TABLE II

Test Equipment (Final Tests)

No.	Item	Source	Description
1	Engine	Wisconsin Motor Corp.	Model ABN .85 HP @ 1400 RPM 2.10 HP @ 1800 RPM
2	Cold Chamber		Cooled by Dry Ice-variable to -70°F. 80 cu. ft.
2A	Cold Air Source for Blowers		Cooled by Dry Ice
3	Flowmeter for Intake Air	Fischer & Porter	Flowrator Size 5 Fig. #735-BH 10" scale
4	Flowmeter for Fuel	Fischer & Porter	Flowrator Size 07 Fig. #130-BH 150 mm scale
4A	Fuel Tank	From Engine	1 gal. cap.
4B	Platform Scale	---	25# by oz.
5	Damper to vent		
6	Damper to test		
7	Exhaust Pressure Safety Valve		set 10" $H_2 0$
8	Exchanger undergoing test		
8A	Crossflow Section		See Fig. 4 for details
8B	Parallel Flow Section		" " " " "
8C	Water Separator		" " " " "

TABLE II (continued)

No.	Item	Source	Description
8D	Float trap		See Fig. 4 for details
9	Air Blower - Crossflow Section	(Torrington Fan (Redmond Motor	-- (5 blades Type O-5527-5 (1/150 HP Model #1-3559
9A	Air Blower - Parallel Flow Section	(Redmond Blower (Century Motor	-- (No. L-3883 (6^V DC 1/80 HP
10	Observation light source	---	Intense light beam across sampling tube outlet. Black background.
11	Not Used		
12	Not Used		
13	Variable Resistance	---	Carbon Pile
14	Not Used		
15	Thermometer - Hot Gas In	Macalaster Bicknell	50° to 920°F
16	Thermometer -- Final Mixture Out	Weston Electr. Inst. Co.	-100° to +100°F
17	Thermometer - Cooling Air In	Weston Electr. Inst. Co.	-100° to +100°F
18	Not Used		
19	Potentiometer	Brown Instr. Div. of Minneapolis Honeywell	0 to 500°F Model #105X4P

TABLE II (concluded)

No.	Item	Source	Description
20	Generator	A & J Auto Ignition Co.	6^V DC 40 amp.
21	Thermometer - Inner Fog Chamber	Weston Electr. Inst. Co.	-100° to +100°F
22	Inner Fog Chamber	---	80 cu. ft. with 12" sq. thermopane window
23	Differential Manometer - Hot Side Pressure Drop	Meriam Inst. Co.	30" H_20
24	Inclined Manometer - Cold Side Pressure Drop	" " "	0 to 4" H_20 by .02"
25	Selector Switch	Brown Instr. Div. of Minneapolis Honeywell	12-pt. #911A1A-34
26	Measure for Condensate	---	200 cc graduate
27	Mixing baffles	---	Sheet Metal
28	Sampling line for exhaust mixture	---	3/8" ID Rubber Tube
29	Duct for full flow of exhaust mixture into cold chamber	---	Sheet Metal 4" Dia.

Date 12/27/51

Run No. 22

SECURITY INFORMATION
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Time	Situation	Cold Chamber °F	Temperatures: Hot Gas: At Engine	In	1/2" Below Inlet	Between Sections: Front	Between Sections: Rear	After Parallel Section	Inside Water Separator	Cooling Air: In	Leaving Crossflow: Top	Leaving Crossflow: Bottom	Leaving Crossflow: Av.	Leaving Parallel Flow Section	Inside Parallel Sec. 6" Before End	Mixture: After Mixing	Mixture: Discharge Duct	Remarks
P.M.			H_0	H_1	H_2	H_3	H_4	H_5	H_6	C_0	C_1	C_2	C_3	C_4	C_5	C_6	C_7	
1:40	Start Engine	-48								--								
1:50		-48	700	435	345	82	83	32	65	-50	--	--	55	20	22		60	
2:00		-48	755	500	410	90	84	40	45	-42	--	--	90	39	40		68	
2:12	Exhaust Bleed	-48	--	--	--	--	--	40	43	--	--	--	--	--	--		--	No Fog
	Into chamber																	" "
2:15		-48	750	525	415	92	98	40	44	-40	--	--	88	32	42		70	" "
2:45	Slowed Down	-48	550	400	290	72	78	40	42	-34	--	--	46	30	40		49	" " Full Exhaust into Ch'br.
3:05		-46	550	--		--	--	40	42	--	--	--	--	30	40		49	" "
3:10	Shut Down	-48	550	405	320	210	205	47	50	0	--	--	105	50	59		--	Heavy Fog (Cooling Blowers Stopped)

Time	Situation	Flows: Engine: Fuel: Flow-meter	Fuel #/Hr.	Fuel Tank: Wt.	Fuel Tank: Diff.	Fuel Tank: Cum. Total	Air: Flow-Meter	Air: CFM	Cooling Air: Crossflow Section CFM	Cooling Air: Parallel Section CFM	Cooling Air: Total CFM	Pressures ("H_2O): Hot Side: Total Δp	Cold Side: Manometer Reading	Cold Side: Zero Reading	Δp	Generator amps	Condensate cc (cum.)
1:40	Start Engine	15-1/2		9#-5 oz.			17-1/2	6.15	12.	24.5	36.5	1"	--	--	--	--	
1:50		--					15-1/2	5.40				1"	--	---	--	30	
2:00		--					15	5.20				1	--	--	[illegible]	40	
2:12	Exh. Bld.	--		8#-10 oz.	0-11 oz	0-11oz.	15-1/2	5.40				1-1/4	--	--	--	--	65
2:15	into Chamber	--					15-1/2	5.40				1	--	--	--	--	
2:45	Slowed Down	13-1/2					10-3/4	3.70				3	--	--	--	38	
3:00		--					10-3/4	3.70				1	--	--	--	5	
3:10	Shut Down	--		7#-12 oz.	0-14	1-9oz	10-3/4	3.70				1	--	--	--	--	213

TABLE III

SECURITY INFORMATION
CONFIDENTIAL

Date 12/30/52

Run No. 24

SECURITY INFORMATION
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TEMPERATURES

Time	Situation	Cold Chamber °F	Hot Gas: At Engine	Hot Gas: In	Hot Gas: 1/2" Below Inlet	Hot Gas: Between Sections, Front	Hot Gas: Between Sections, Rear	Hot Gas: After Parallel Section	Hot Gas: Inside Water Separator	Cooling Air: In	Cooling Air: Leaving Crossflow, Top	Cooling Air: Leaving Crossflow, Bottom	Cooling Air: Leaving Crossflow, Av.	Cooling Air: Leaving Parallel Flow Section	Cooling Air: Inside Parallel Sec. 6" Before End	Mixture: After Mixing	Mixture: Discharge Duct	Remarks
P.M.			H_0	H_1	H_2	H_3	H_4	H_5	H_6	C_0	C_1	C_2	C_3	C_4	C_5	C_6	C_7	
2:15	Start Engine	-70	75	75	--	--	--	--	--	--	--	--	--	--	--	--	--	
2:45	Start Readings	-70	840	490	--	--	--	--	--	70	--	--	--	--	--	--	--	
2:50	Exh. Bleed into Chbr.	-70	910	550	225	80	82	40	45	-40	65	55	60	30	37	48	--	No Fog
2:55	Full Exh. into Chbr.	-70	920	550	270	90	90	40	45	-41	75	68	77	35	45	52	--	" "
3:00	Exh. Bled into Chbr.	-68	900	500	238	70	70	40	44	-64	44	44	38	26	42	45	80	" "
3:05	"	-70	900	510	255	85	76	45	44	-44	52	55	50	40	50	51	--	" "
3:10	"	-70	890	510	255	85	76	45	43	-44	52	55	50	35	44	49	--	Full Exhaust into Chamber No Fog
3:15	"	-70	900	510	255	84	75	45	42	-43	50	55	50	35	44	49	--	" "
3:40	Shut Down		880	500	250	83	75	45	42	-41	50	55	50	40	47	50	--	" "

FLOWS / PRESSURES ("H_2O)

Time	Situation	Engine Fuel: Flow-meter	Engine Fuel: Fuel #/Hr.	Engine Fuel Tank: Wt.	Engine Fuel Tank: Diff.	Engine Fuel Tank: Cum. Total	Engine Air: Flow-Meter	Engine Air: CFM	Cooling Air: Crossflow Section CFM	Cooling Air: Parallel Section CFM	Cooling Air: Total CFM	Hot Side: Total Δ p	Cold Side: Manometer Reading	Cold Side: Zero Reading	Δ p	Generator amps	Condensate cc. (cum.)
2:15	Start Engine	--	--	9#-8 oz.	--	--	--	--	--	--	--	--	--	0.70	--	--	
2:45	Start Readings	12-1/4	--	--	--	--	13-1/4	4.60	12.0	16.0	28.0	1	1.07	.70	0.37	31	
2:50	Exh. Bleed into Chbr.	15-1/4	[illegible]	--	--	--	17-1/2	6.15	12.0	17.5	29.5	3/4	1.12	.70	.42	33	
2:55	Full Exh. into Chbr.	12-1/2	--	--	--	--	10-1/4	3.55	12.0	25.8	37.8	1-1/4	1.52	.70	.82	25	
3:00	Exh. Bleed into Chbr.	12-1/2	--	--	--	--	15-1/2	5.40	12.0	24.6	36.6	1-1/4	1.47	.70	.77	30	
3:05	"	12-1/2	--	--	--	--	15-1/2	5.40	12.0	23.8	35.8	1-1/4	1.42	.70	.72	30	
3:10	"	12-1/2	--	--	--	--	15-1/2	5.40	12.0	23.8	35.8	1-1/4	1.42	.70	.72	30	
3:15	"	14	1.12	8#-6 oz.	1#-2 oz	1#-2oz.	15	5.20	12.0	23.8	35.8	1	1.42	.70	.72	30	154
3:40	Shut Down	10	0.75	8#-1 oz.	0#-5 oz	1#-7oz.	--	--	12.0	24.1	36.1	3/4	1.44	.70	.74	32	200

TABLE IV

SECURITY INFORMATION
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APPENDIX B

DESIGN CALCULATIONS

Nomenclature

A	-	total heat transfer area
A_c	-	core area of section (free flow area)
A_p	-	heat transfer area of flat plates
A_f	-	heat transfer area of fins
c_p	-	heat capacity
E	-	fin efficiency
f	-	friction factor (from Fanning Equation)
G	-	mass rate of flow per unit area
g	-	gravitational constant
h	-	film heat transfer coefficient
K_c	-	contraction coefficient
K_e	-	expansion coefficient
Δp	-	pressure loss
Q	-	heat flow
R_e	-	Reynolds number = $4rG/\mu$
r	-	hydraulic radius, $4r$ = 0.01012 ft. for "A" section
Δt	-	temperature difference
Δt_m	-	mean temperature difference for heat flow
Δt_l	-	logarithmic mean temperature difference
U	-	over-all heat transfer coefficient
W	-	mass flow rate

Subscripts

a	-	pertaining to air side
e	-	pertaining to exhaust side

Greek Letters

μ	-	viscosity
ρ	-	density
σ	-	ratio of free flow (core) area to total frontal area

Requirements: Cool 17.9 #/hr of exhaust gas (8.7% by wgt. water) from 1000° F to 40° F using not more than about 90 #/hr of air available at -40° F. The pressure drop on the exhaust side can be up to 2 psi but the air-side pressure drop should be kept to a minimum. To prevent the condensed water from freezing the cooling will be done in two sections: a cross-flow exchanger to cool the exhaust from 1000° to 300° F; and a parallel-flow section to cool the exhaust from 300° to 40° F and also condense out most of the water in the exhaust.

Design of Exchanger I

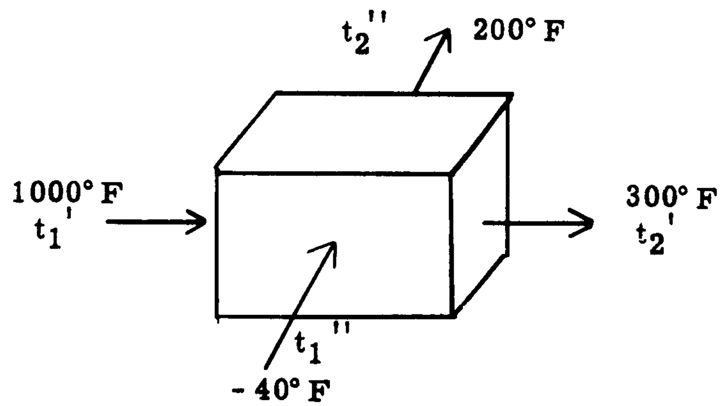
Heat load: $Q = Wc_p \Delta t$

$= 17.9\,(0.25)(1000-300) = 3130$ Btu/hr

For air side: $3130 = (t_2 + 40)(0.25)w$

$w = 12520/(t_2 + 40)$

Assume $t_2 = 200°$ F, then $w = 52.2$ #/hr of air



Calculation of mean temperature difference - see McAdams, p. 147, Chart G.

$$Y = \Delta t_m / \Delta t_1 = f\,(X,\ Z)$$

$$\Delta t_1 = \frac{(t_1' - t_2'') - (t_2' - t_1'')}{\ln \dfrac{t_1' - t_2''}{t_2' - t_1''}} = \frac{940}{2.34} = 402° \text{F}$$

$$X = \frac{t_2'' - t_1''}{t_1' - t_1''} = \frac{240}{1040} = 0.230$$

$$Z = \frac{t_1' - t_2'}{t_2'' - t_1''} = \frac{700}{240} = 2.92$$

$$Y = 0.92$$

$$\Delta t_m = 0.92\ (402) = 370°\,F$$

For heat transfer

$$Q = UA\Delta t_m = 3130 \text{ Btu/hr}$$

$$UA = 3130/370 = 8.46$$

Summing up heat transfer resistance:

$$\frac{1}{UA} = \left(\frac{1}{hAE}\right)_a + \text{wall resistance} + \left(\frac{1}{hAE}\right)_e$$

The wall resistance will be negligible and E, the fin efficiency, will be about 0.95.

For the first approximation assume that $Eh_a = Eh_e = 6.2$ Btu/hr - ft^2 - °F
Thus $U = \frac{1}{2}\ hE = 3.1$
A = 8.46/3.1 = 2.73 sq. ft. (approximately)

If exchanger is 4 1/2" by 1 1/2" by 10 layers then for air side

$$A_c = 0.25\ (4.5)\ (10)\ /144 = 0.0782 \text{ sq. ft.}$$

$$A_p = 1.5\ (4.5)\ (18)\ /144 = 0.844 \text{ sq. ft}$$

$$A_f = 0.25\ (1.5)\ (2)\ (11.1)\ (4.5)\ (10)/144 = 2.60 \text{ sq. ft.}$$

$$A = 2.60 + 0.844 = 3.44$$

For exhaust side:

$$A_c = 9\ (0.25)\ (1.5)/144 = 0.0234 \text{ sq. ft.}$$

$$A_p = 1.5\ (4.5)\ (18)/144 = 0.844 \text{ sq. ft.}$$

$$A_f = 0.25\ (4.5)\ (2)\ (11.1)\ (1.5)\ (9)/144 = 2.34 \text{ sq. ft.}$$

$$A = 2.34 + 0.844 = 3.18 \text{ sq. ft.}$$

to check the performance of this section:

For exhaust side:

$$G = w/A_c = 17.9/(0.0234) = 765 \text{ \#/hr.- sq. ft.}$$

$$R_e = 4rG/\mu = \frac{(0.01012)\ (765)}{(0.03)\ (2.42)} = 107$$

From Fig. 9 of Navy Report

$$(h/Gc_p)\ (Pr)^{2/3} = 0.027$$

$$h = Gc_p\ (0.027)/Pr^{2/3} = \frac{765\ (0.25)\ (0.027)}{0.82}$$

$$= 6.3 \text{ Btu/hr.-sq. ft. °F}$$

For air side, w = 52.2, using the same expressions except that $\mu = 0.017(2.42)$

$$G = 669 \text{ \#/hr.-ft.}^2$$

$$Re = 165$$

$$h = 4.07 \text{ Btu/hr.-sq. ft.-°F}$$

$$\frac{1}{U_a} = \frac{1}{h_a E} + \frac{A_a}{hA_e E} + \text{negligible wall resistance}$$

$$= \frac{1}{4.07(0.95)} + \frac{3.44}{6.3(0.95)(3.18)} = 0.4395$$

$$U_a = 2.27 \text{ Btu/hr.-sq. ft.-}^\circ\text{F}$$

$$Q = UA\Delta t_m = 2.27(3.44)(370) = 2890 \text{ Btu/hr.}$$

Since it is necessary to transfer 3130 Btu/hr. a larger section is needed.

Revised Design

For the second trial the following exchanger dimensions will be used:

Length	4.75"
Depth	1.5"
Layers for Air	10
Layers for Exhaust	9

For air side

$$A_c = 0.0782\left(\frac{4.75}{4.5}\right) = 0.0824 \text{ sq. ft.}$$

$$A_p = 0.844\left(\frac{4.75}{4.5}\right) = 0.89 \text{ sq. ft.}$$

$$A_f = 2.60\left(\frac{4.75}{4.5}\right) = 2.74 \text{ sq. ft.}$$

$$A = 2.74 + 0.89 = 3.63 \text{ sq. ft.}$$

For exhaust side:

$$A_c = 0.0234\left(\frac{4.75}{4.5}\right) = 0.0247 \text{ sq. ft.}$$

$$A_p = 0.844\left(\frac{4.75}{4.5}\right) = 0.89 \text{ sq. ft.}$$

$$A_f = 2.34\,(4.75/4.5) = 2.47 \text{ sq. ft.}$$

$$A = 2.47 + 0.89 = 3.36 \text{ sq. ft.}$$

For exhaust

$$G = {}^{17.9}/0.0247 = 725 \text{ \#/hr.-sq. ft.}$$

$$Re = \frac{(0.01012)(725)}{0.03(2.42)} = 101$$

$$h = \frac{0.028(725)\,(0.25)}{0.82} = 6.2 \text{ Btu/hr.-ft.}^2\,{}^\circ\text{F}$$

For air

$$G = {}^{52.2}/0.0824 = 634 \ \#/\text{hr. ft.}^2$$

$$Re = \frac{(0.01012)(634)}{(0.017)(2.42)} = 156$$

$$h = \frac{0.0208\ (634)\ (0.25)}{0.82} = 4.03 \text{ Btu/hr. ft.}^2 \ ^\circ F$$

Since $E = 1.0$ for low values of U

$$\frac{1}{U} = \frac{1}{4.03} + \frac{3.63}{3.36\ (6.2)} = 0.423$$

$$U = 2.37 \text{ Btu/hr.-sq. ft.-}^\circ F$$

$$Q = UA\Delta t_m = 2.37\ (3.63)\ (370)$$

$$= 3180 \text{ Btu/hr.}$$

thus this design is satisfactory.

Air Pressure Drop Through Exchanger

$$\text{Total } \Delta p = \text{entrance loss + core loss + exit loss}$$

$$= \Delta p_i + \Delta p_c + \Delta p_o$$

$$(\Delta p/\rho_1)\,i = K_c \frac{G^2}{2g\rho^2} + \frac{G^2}{2g\rho_1{}^2}\,(1 - \sigma^2)$$

$$(\Delta p/\rho_1)\,c = \frac{G^2}{2g\rho_1{}^2}\left[2\left(\frac{\rho_1}{\rho_2} - 1\right) + f\,\frac{A}{A_c}\,\frac{\rho_1}{\rho_{av}}\right]$$

$$(\Delta p/\rho_2)_o = K_e \frac{G^2}{2g\rho_2{}^2} - \frac{G^2}{2g\rho_2{}^2}\,(1 - \sigma^2)$$

$$K_e = (1 - \sigma)^2$$

let $\rho^* = \rho_1/\rho_2$, then $\rho_{av} = \dfrac{\rho_1 + \rho_2}{2} = \dfrac{\rho_1}{2}\left(\dfrac{\rho^* + 1}{\rho^*}\right)$

After adding the expressions for the individual pressure drops and expressing all densities in terms of ρ^* and ρ_1,

$$\Delta_p = \frac{G^2}{2g\rho_1}\left[K_c - 1 - \sigma^2 + 2\rho^*\left\{\frac{\sigma^2 + 1 + K_e}{2} + \frac{A}{Ac}\frac{f}{(1 + \rho^*)}\right\}\right]$$

Air pressure drop:

σ = A_c/Frontal area = $\dfrac{11.85}{4.75 \times 4.8}$ = 0.52

σ^2 = 0.274

K_e = $(1 - \sigma)^2$ = 0.230

K_c = 0.34

ρ^* = t^2/t^1 = $680/420$ = 1.62

ρ_1 = $\dfrac{39}{359}\left(\dfrac{492}{420}\right)$ = 0.0946 #/cu. ft.

A/A_c = $2.74/0.0824$ = 33.3

Re = 156, f = 0.09

G = 634

$$\Delta p = \frac{634^2}{8.34 \times 10^8(0.0946)}\left[-0.934 + 3.24\left\{0.752 + \frac{33.3(0.09)}{2.62}\right\}\right]$$

$= 0.0266$ #/sq. ft.

or $0.0266(12)/62.4 = 0.0051$ inches of water

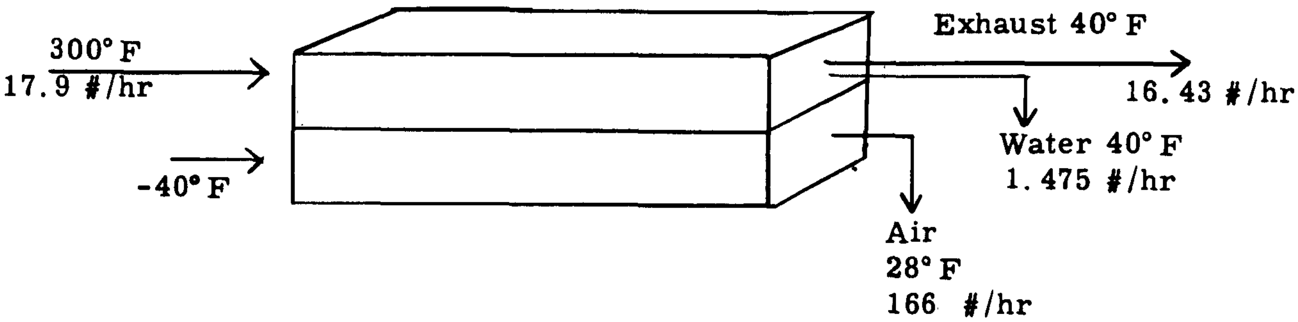
Design of Exchanger II

Composition of Exhaust

	Total	17.90
8.7%	Water	1.56
	Dry gas	16.34 lb/hr.

at 40° F saturated air contains 0.0052 lb. water per lb. dry air.

Water out in vapor = 16.34 (0.0052) = 0.085
Water condensed = 1.56 - 0.085 = 1.475
Gas out = 16.34 + 0.085 = 16.425



Heat Load on Condenser

Latent heat of water = 1071 Btu/# at 40° F

$$Q_{water} = 1.475\ (1071) + 1.475\ (300 - 40)\ (0.44)$$
$$= 1751 \text{ Btu/hr.}$$

$$Q_{gas} = 16.43\ (0.25)\ (300 - 40)$$
$$= 1068 \text{ Btu/hr.}$$

$$Q = 1068 + 1751 = 2819 \text{ Btu/hr.}$$

$$W_a = Q/cp\ \Delta t = 2819/0.25\ (28 + 40) = 166 \text{ \#/hr.}$$

Mean Temperature Difference

$$\Delta t_m = \Delta t_l = \frac{(300 + 40) - (40 - 28)}{\ln \frac{340}{12}} = 98°F$$

$Q = UA\Delta t_m$

$UA = 2819/98 = 28.8$

if U is about 2 then A = 14.4 sq. ft.

Assume the following dimensions for the end face of the exchanger

Width:	3"
Layers for air flow	6
Layers for exhaust flow	7
Height:	3 1/4" (approximate)

Air side areas

$A_c = (0.25)(3)(6)/144 = 0.0313$ sq. ft.

per inch of length:

$A_p = 3(1)(12) = 36$ sq. in.

$A_f = (0.25)(1)(2)(11.1)(3)(6) = 99.6$ sq. in.

$A = 36 + 99.6 = 135.6$ sq. in.

Exhaust side areas

$A_c = (0.25)(3)(7)/144 = 0.0365$ sq. ft.

per inch of length

$A_p = 3(1)(12) = 36$ sq. in.

$A_f = (0.25)(1)(2)(11.1)(3)(7) = 116.3$ sq. in.

$A = 36 + 116.3 = 152.3$ sq. in.

For air:

$$G = {}^{166}/0.0313 = 5310 \text{ \# hr.-sq. ft.}$$

$$Re = \frac{(0.01012)(5310)}{(0.016)\ (2.42)} = 1390$$

$$\frac{h}{Gc_p} \quad (Pr)^{2/3} = 0.0047$$

$$h = \frac{(0.0047)(5310)(0.25)}{(0.82)} = 7.61 \ {}^{Btu}/\text{hr. ft.}^2\ {}^\circ F$$

For Exhaust:

$$G = 17.9/0.0365 = 491 \text{ \#/hr.-sq. ft.}$$

$$Re = \frac{(0.0102)(491)}{(0.018)(2.42)} = 114$$

$$\left(\frac{h}{Gc_p}\right)\left(Pr\right)^{2/3} = 0.0255$$

$$h = \frac{(0.0255)\ (491)(0.25)}{0.82} = 3.82 \ {}^{Btu}/\text{hr. sq. ft.} {}^\circ F$$

$$\frac{1}{Ua} = \frac{1}{h_a} + \frac{A_a}{h_e A_c} = \frac{1}{7.61} + \frac{135.6}{3.82(152.3)} = 0.364$$

$$U_a = 2.75 \text{ Btu/hr. - ft.}^2 - {}^\circ F$$

$$A_a = \frac{Q}{U \Delta t_m} = \frac{2819}{(2.75)(98)} = 10.45 \text{ sq. ft.}$$

$$\text{Length of exchanger} = \frac{10.45\ (144)}{135.6} = 11.1 \text{ in.}$$

For safety factor make length 12 inches, then areas are as follows:

For air:

$$A_c = 0.0313 \text{ sq. ft.}$$

$$A = 135.6\,(12)/144 = 11.3 \text{ sq. ft.}$$

For exhaust:

$$A_c = 0.0365 \text{ sq. ft.}$$

$$A = 152.3\,(12)/144 = 12.7 \text{ sq. ft.}$$

Air Pressure Drop

Using the same expression for the pressure drop as was used for Exchanger I, the quantities involved are:

$$A/A_c = {}^{11.3}/0.0313 = 361$$

$$Re = 1390\ ,\ f = 0.019$$

$$\rho^* = \rho_1/\rho_2 = \frac{P_1\,T_2}{P_2\,T_1} = \frac{488}{420} = 1.16$$

$$\rho_1 = 0.0946 \ \#/\text{cu. ft.}$$

$$\Delta p = \frac{(5310)^2}{8.34 \times 10^8 (0.0946)} \left\{ -0.867 + 2.32 \left[0.752 + \frac{361(0.019)}{1 + 1.16} \right] \right\}$$

$$= 2.95 \ \#/\ \text{sq. ft.}$$

$$\text{inches of water} = 2.95\,(12)/_{62.4} = 0.568$$